The Association
of
Engineering and Shipbuilding
Draughtsmen.

VITAL FACTORS IN LOCOMOTIVE DESIGN.

By GEORGE W. McARD.

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Before a locomotive can be designed with any degree of assurance that the resulting machine will do all that is required of it, much investigation is involved and many enquiries are to be satisfied. Some constructors go so far as to issue a questionnaire to a prospective client and it is proposed, in the notes which follow, to consider the various points in sequential order as they affect the locomotive-to-be. Before any design can proceed satisfactorily, however, definite information is required covering—

1.—Gauge of railway, actual or proposed.
2.—Weight of rails in lbs. per yard.
3.—Longest non-stop journey engine is to make.
4.—Distance between watering stations.
5.—Fuel to be used.
6.—Radius of sharpest curve—(*a*) on main line, (*b*) in shunting yards.
7.—Gradients—length and steepest inclination of same, and if combined with a curve, what is the curve radius ; also whether grade is compensated.
8.—Running schedules, with line profile and grades combined.
9.—Maximum load in tons to be hauled, and character of stock used.
10.—Particulars of buffing and draw-gear.
11.—Loading gauge for line.
12.—Diameter of turntables and total permissible wheelbase for engine and tender.

Track.—Usually this is already in existence, and attention must therefore be devoted to the road upon which the engine will operate. Where, however, a new track is proposed, the following notes concern the problems met when considering the gauge and other features which affect the initial cost of the track and that involved in its operation.

The number of gauges in service throughout the world to-day, ignoring model railways, is approximately forty, commencing with the 18″ gauge used for quarry tracks and similar purposes, and rising to a maximum of 5′-6″. They can, however, all be classified broadly under three heads :—

(1)—Narrow, represented by the 2′-6″, metre and 3′-6″ lines.
(2)—Standard, ,, ,, 4′-8½″ system, and
(3)—Broad, ,, ,, 5′-6″ track.

No attempt at standardisation ever appears to have been made, and to-day the problem is much too complex to permit of such a step, involving locomotives, coaching and wagon stock, and track in the changes that would be necessary. The Australian railways operate on three main gauges, viz., 5′-3″, 4′-8½″ and 3′-6″, and it is proposed that the 4′-8½″ shall be the ultimate standard for that country. To this end, new locomotive stock for the 5′-3″ sections is designed in such manner as will simplify its ultimate conversion to the narrower gauge as far as possible, and this can be accomplished quite simply by building the whole engine, except wheels and brake crossbeams, for the 4′-8½″ gauge and "stepping out" the wheel rims and tyres to suit the larger gauge. On conversion, a new set of wheels will suffice to effect the change, using the original axles if still in good condition, shortening the distance over shoulders of the brake beams and moving the brake blocks and hangers nearer the frames in order to retain their position correctly relative to the tyres.

Possibly the African Railways can boast the greatest care in planning for ultimate interconnection with other lines, as most of the systems in the African continent are built to the 3′-6″ gauge, and possess a generous load gauge.

The standard gauge of 4′-8½″ is employed almost universally in Great Britain, except for small self-contained systems in quarries and similar workings. It is also the gauge adopted on the continent (except France, Spain, Portugal and Russia), in the U.S.A., Canada, China and several other important countries; and an examination, however cursory, of the latest heavy American engines will suffice to show that this gauge will carry any power unit that may be required, always assuming the track—rail sections, sleepers and ballast—are sufficiently strong to sustain the load, and the loading gauge generous enough.

Under the heads of "Narrow" and "Broad" are placed, respectively, all tracks under and over 4′-8½″. As already indicated, these are too numerous to quote individually, but the former comprise the 2′-0″, 2′-6″, 3′-0″, metre and 3′-6″, while the "Broad" gauges embrace the 5′-0″, 5′-3″ and 5′-6″ systems. The original Great Western Railway broad gauge of 7′-0″ is now practically non-existent.

The standard gauge, although excellent in many respects, would not be the most suitable for terrain involving numerous and fairly sharp curves, and in this class of country a narrow gauge provides the only satisfactory solution, although, as will be seen, at an increased annual cost for the running and maintenance of its stock. The following are the advantages and disadvantages of the narrow gauge, the extent of each being more or less proportionate to the difference between the gauge and the standard gauge.

Advantages—

(*a*) Reduced initial outlay on land and rolling stock.
(*b*) Fewer tunnels, cuttings and embankments, and consequently cheaper cost of track per mile.
(*c*) Less time required in construction.

Disadvantages—

(*a*) In trying to follow the contour of the country under survey, the line is almost invariably abundant in curves and heavy grades, which reduce the available engine drawbar pull considerably. Hence, whilst an extra capital expenditure is saved, an additional annual charge is placed on locomotive power expenses which sooner or later outbalances the capital saved.
(*b*) Greater transverse overhang required, with consequent increased liability to derail. In England, with a gauge of 4′-8½″, the maximum width of moving structure is approximately 8′-8″. On one of the South American Railways—the Bolivian Railway—with a gauge of 3′-3⅜″—1′-5⅛″ less than the former—the maximum width is 9′-6″—nearly a foot more, making in all an extra overhang of tanks, etc., of more than two feet. (Admittedly, the centre of gravity and the speed are reduced to compensate for this, but many engineers question the policy of such methods where a wider gauge might originally have been chosen).
(*c*) Greatly reduced speeds (hence, smaller wheels).
(*d*) Greater consumption of lubricating oil per mile, due to smaller wheels involved. In the heavy locomotive stock in particular this has been very pronounced and is considered to be in part due to the closer proximity to the track of the moving parts, dust and grit the more easily obtaining access to journals. (A pair of journals, one 8″ dia. × 9″ long on a standard gauge engine, the other 8″ dia. × 7″ long on a narrow gauge engine, may be compared to prove this statement *re* greater oil consumption. Assume wheels of 6′-9″ dia. and 4′-0″ dia. respectively and compare journal surface swept through per mile, with equivalent loads per square inch in each case).
(*e*) Inconvenience and serious losses owing to break of gauge when coming in contact with line of standard gauge.
(*f*) A necessary cramping of parts accompanied by reduced accessibility for repairs, adjustments, etc.
(*g*) Inside cylinders almost impossible.

Rail Weight.—Reference has already been made to the quality of the track, with the implication that a wide gauge does not

necessarily presuppose more powerful locomotive stock operating thereon. Actually instances can be quoted where a narrow gauge employs more powerful engines than certain lines having a bigger gauge—the ruling factor here is the standard of excellence of the track and the bridges included in the system. The weight of the rails, in pounds per lineal yard, also determines the permissible maximum load in tons which any axle of a two-cylinder engine may carry, and the following ratios have been found to give safe results in British rail practice.

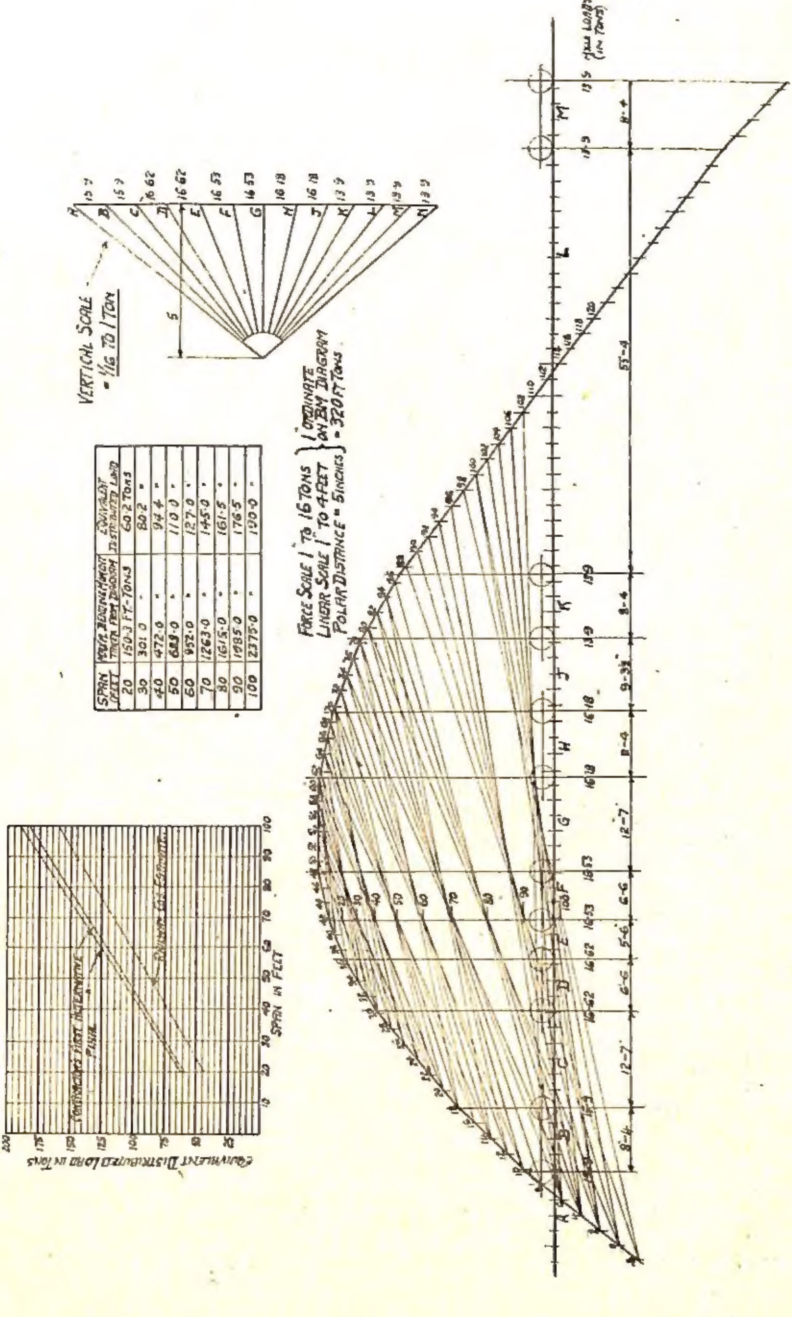
Main lines.—Axle loading (in tons) may be 1/5th rail weight (in lbs. per yard).

Secondary lines.—Axle loading (in tons) may be 1/6th rail weight to compensate for a track inferior in ballasting, general finish and maintenance.

Bridges.—The value of 5 to 1 given above is occasionally transgressed for special engines—three- or four-cylinder types—where the balancing is of such an order that with a higher static axle loading the pounding of track and bridges is less severe than with the heavy two-cylinder engines operating on that section. This is usually decided by reference to the Permanent Way Engineer's department and collaboration with those concerned in the maintenance of the track in safe running order, a diagram on the lines of Fig. 1 being prepared by the locomotive designer for the particular engine and the bridges involved. (For detailed information as to the method employed in producing this diagram, see Mr. Clayton's paper on "The Bridge Curve" in *Journal* No. 27 of the Institution of Loco. Engineers). Fig. 1 shows the bridge curve as generated for the Diesel electric power houses built in 1932 by Armstrong Whitworth for the B.A. Gt. S. Railway.

Turntables.—Another factor which affects new locomotive stock where this reaches a big overall length is the size of turntables on the system, and on account of this it may be necessary to adopt special measures to enable a high power unit to be built within a shorter total wheelbase than would otherwise be adopted. The omission of the trailing truck, resulting in, say, a 4-6-0 type instead of a 4-6-2 would offer a solution that would probably meet the case so far as the turntable diameter is concerned, but might render the engine designer's task more difficult in obtaining the requisite boiler power.

An alternative that appears to the writer to offer better possibilities is that shown at B in Fig. 2, where the tender is carried on two bogies, the leading unit serving to support the locomotive rear end and the tender front end by the use of articulated centre castings. (A 4-8-2 type engine of normal design is shown at A by way of comparison). This permits the use of a wide firebox



SPAN (FEET)	MAX'M BENDING MOMENT TAKEN FROM DIAGRAM	EQUIVALENT DISTRIBUTED LOAD
20	150·3 FT-TONS	60·2 TONS
30	301·0 "	80·2 "
40	472·0 "	94·4 "
50	688·0 "	110·0 "
60	952·0 "	127·0 "
70	1263·0 "	145·0 "
80	1615·0 "	161·5 "
90	1985·0 "	176·5 "
100	2375·0 "	190·0 "

Fig. 1.

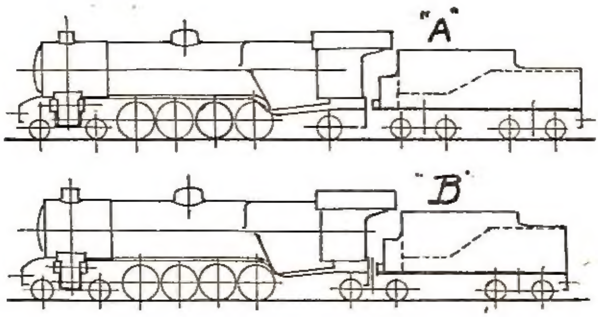


Fig. 2.

and incidentally cuts out the relative movement vertically of the two frames with its attendant problems. The L.N.E. Railway adapted a locomotive to this design some years ago, but the scheme apparently failed to take advantage of the full benefits that are possible in such a design, as it does not appear to have been repeated.

Maximum Load Gauge.—As locomotives do not function entirely in open country, but frequently pass through cuttings, arches, tunnels, and in proximity to obstructions such as platforms, signals, etc., some limitation must be made to their contour as observed in end view. This line of demarcation when fixed is known as the maximum loading gauge, and, unfortunately, varies almost to the same degree as does the track gauge. Even in Great Britain, almost every one of the individual lines existing prior to the grouping of the four main lines had a separate load gauge. Although this has been combined to a large extent to afford a common limit for certain new stock, it is not financially possible to alter all structures over any existing system to one common gauge, and many of the old limitations therefore still stand. Foreign lines are in much the same condition, each having its own particular loading gauge, and it is imperative that every new design be planned to pass the specified limit. Inspecting engineers who know their job will insist on a gauge being provided by the engine builder with which to test the first of any new engines, assuming, of course, that the design indicates that full use has been made of the limits of the gauge.

This restriction can give the designer of a heavy engine much anxiety in providing a big-boilered unit having large driving wheels, while yet keeping his machine within the gauge. The driving wheels force the boiler centre up and the large heating surfaces required compel the adoption of a big barrel diameter with the result that not only does the chimney suffer severe attenuation,

but the safety valves may need redesigning to suit the shortened height available, and the cab demands special treatment where it passes over the firebox.

Gradients.—Although the narrower gauge lines tend to include gradients more in number and even greater in severity than those found on the wider gauges, a limit exists beyond which it is not economical to proceed except by the use of a rack system over the more difficult section. Grades of 1 in 16 are found in service to be the aforesaid limit, and such are only used, as a general rule, where the driver can approach at fairly high speed—and, therefore, with considerable kinetic energy in hand—crossing the summit at an appreciably reduced speed. A glance at Fig. 3 will show that two adverse influences act against the forward movement of a locomotive and train up an incline, first, the pull of gravity, and second, the reduced value of the adhesive load ; this latter, however, is more theoretical than actual.

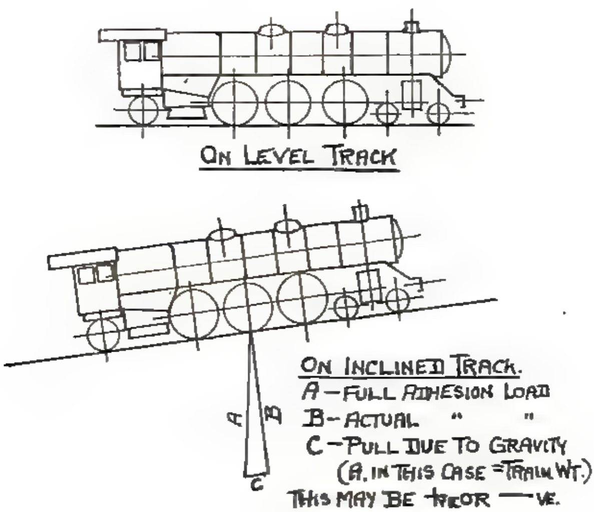


Fig. 3.

Where it is not possible to cross a section of country by a direct line using grades of ordinary values for adhesion engines, and a tunnel would be of a length well-nigh impossible to construct or operate, several methods may be adopted, the first being to zigzag the track up the hillside by a carefully planned road in the form of a pass, and on reaching a suitable elevation tunnel through to the opposite side. Alternatively, a rack line may continue the route over the range from the point where the adhesion line ends, as the Transandine Railway in South America.

An alternative to the rack system is the cable, operated by a stationary hauling engine at the top of the bank, the latter taking the train load, while the locomotive power is mostly absorbed in transporting itself to the summit. Trains to Edinburgh from the Queen Street (Glasgow) terminus of the old North British Railway were formerly assisted up the grade to Cowlairs by cable, and those from San Paulo (Brazil) still use similar means for reaching the higher level of the Sierras.

Curves.—The influence of curves on a new locomotive design is felt in several directions, and where these are particularly sharp, the whole design—general and detail—will be affected. On the standard and broad gauges the effect is less pronounced than on the narrow gauge lines, but in some of these cases it is very marked. Not only does the passage of the vehicle round a curve demand an increase of power to overcome the extra resistance—unless a reduction in speed is permissible to compensate—but freedom of the several fixed wheels and axles as well as the bogie and truck in any position must be assured without imposing any undue strain on the vehicle framing.

All new designs are subjected to a curve test in the design room to ensure that the rigid wheelbase is suitable for the sharpest curve, as well as to ascertain the side movements required for bogie and truck. Where sharp curves exist, it may be necessary to thin some of the coupled wheel tyre flanges, or even to remove the entire flange, and possibly allow a small clearance between some of the coupled wheel bosses and the axle-box faces and between the box flanges and guide faces. To test for freedom an engine may be laid down on an eighth scale drawing of the track with the "spread" of rails included, or by the Roy process, which is superior to the older method in that it permits the exact measurement of all the clearances required.

The following explanation of the Roy method will assist the designer who is unfamiliar with this test, noting at the outset that the flange on the outer wheel of the leading fixed axle—usually in the coupled group—tends to make contact with the rail. The trailing fixed axle also tends to take up a radial position on the curve and not infrequently, where four or more axles are coupled, the flange of the wheel on the third axle contacts the inner rail before those which follow, unless the flanges of the third pair of wheels are thinned or removed altogether. Fig. 4 shows the method now to be described, the example chosen being a 2-8-0 type locomotive, when running forward.

Let a_1 a_2 a_3 a_4 = wheelbase.
r = length of leading truck arm.
R = curve radius.

s = half the total play between wheel flanges and rail heads, plotted outward from track centre line.

e = half the total play between wheel flanges and rail heads plus amount of gauge widening plotted inwards from track centre line.

A_1 = maximum possible radial displacement of truck centre.

A_5 = maximum side movement available on trailing coupled axle, *i.e.*, between axlebox and wheelboss plus that between axlebox flange and guide face.

a = striking angle (out of scale).

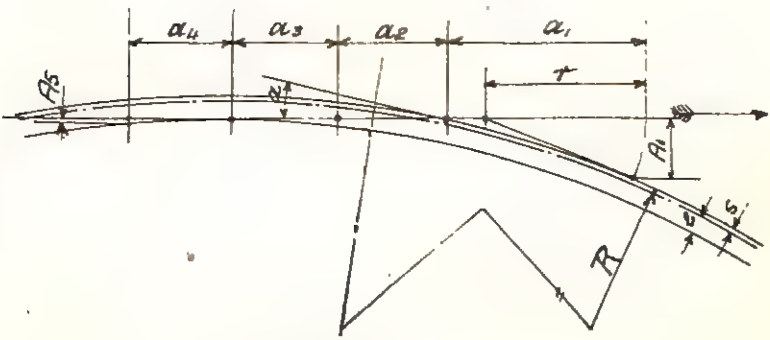


Fig. 4.

The method to be followed is well illustrated in the above diagram, the main points to be watched being the scales to which the curve and the engine wheelbase are to be drawn (see next paragraph) and the positioning of the engine on the curve.

If the transverse scale (comprising wheel flange play, gauge widening and side play of axles) is $1 : b$ and the longitudinal scale (wheelbase) $1 : bn$, then the curve radius R must be drawn to the scale $1 : bn^2$.

TABLE 1.

Transverse Scale		$1 : b$	1 : 1	1 : 2	1 : 4
Longitudinal Scale for	$n = 5$	$1 : bn$	1 : 5	1 : 10	1 : 20
	$n = 10$	$1 : bn$	1 : 10	1 : 20	1 : 40
	$n = 12$	$1 : bn$	1 : 12	1 : 24	1 : 48
	$n = 20$	$1 : bn$	1 : 20	1 : 40	1 : 80
Scale of Curve Radius "R" for	$n = 5$	$1 : bn^2$	1 : 25	1 : 50	1 : 100
	$n = 10$	$1 : bn^2$	1 : 100	1 : 200	1 : 400
	$n = 12$	$1 : bn^2$	1 : 144	1 : 288	1 : 576
	$n = 20$	$1 : bn^2$	1 : 400	1 : 800	1 : 1600

Roy's method is satisfactory for curves not sharper than 180 metres (590 feet), providing the value of n does not exceed 10. In all other cases it gives excessive side displacement, and Vogel's method should be employed.

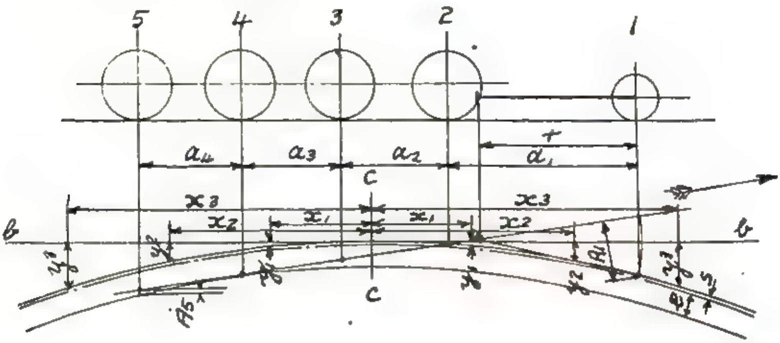


Fig. 5.

The track curve in this case, according to Henschel (to whom the author is also indebted for the foregoing description of Roy's method), is plotted from the horizontal base-line b - b, with the x and y quantities as given in Table 2. All transverse measurements are drawn vertically and all longitudinal measurements parallel to the base-line b - b, and not to the axis of the vehicle. Contrary to Roy's method, the scale to which longitudinal distances are drawn is independent of the scale used for the track curve.

With 1 : b as the scale of transverse distances, the ordinates of the axis of the track are expressed by the formula.

$$y = l^2/2\,R\,b \text{ (in millimetres)} \qquad (1)$$

where l = the real distance of the point on the track centre line from the radius C - C in millimetres.

R = curve radius in millimetres.

To obtain the abscissae x, draw the real distance l to any scale 1 : bn.

The object of providing freedom of movement of the vehicle on the curve is partly to reduce friction to the smallest dimensions, and partly to avoid straining the frame structure. Main line curves are always easier than siding curves in order that the train may traverse them at moderately high speed, but whichever curve the vehicle passes through should be traversed freely and without forcing the frame at any point.

Not only must the frame and undergear respond favourably to any curve condition, but the draw-gear must be equally free to haul in a straight line, and unless this point is carefully watched, the gear may be loaded in a manner for which it has not been designed—by jamming against the edge of the opening in the buffer

TABLE 2.

ABSCISSAE AND ORDINATES FOR $b_m = 40$
R = CURVE RADIUS IN MILLIMETRES $b = 4$ AND $n = 10$ OR $b = 2$ AND $n = 20$

Abscissae in mm		Test Distance 1.1 l	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000	11000	12000	13000	14000	15000	16000
		Distance Scale $x = \frac{l}{b_m} = \frac{l}{40}$	25	50	75	100	125	150	175	200	225	250	275	300	325	350	375	400
Ordinates $y = l^2/2R.b$ in mm	R = 300000	$\frac{1}{b} = \frac{1}{2}$	0·83	3·33	7·5	13 3	20·8	30·0	40·8	53·3	67 5	83 3	101·0	120·0	140·8	163·2	187·5	213 0
		$\frac{1}{b} = \frac{1}{4}$	0 42	1·67	3·75	6·67	10·4	15·0	20·4	26·67	33·7	41·6	50·5	60 0	70·5	81·7	93·8	106·6
	R = 250000	$\frac{1}{b} = \frac{1}{2}$	1·00	4·00	9·00	16·0	25·0	36·0	49 0	64 0	81 0	100·0	121 0	144·0	169·0	196·0	225·0	256·0
		$\frac{1}{b} = \frac{1}{4}$	0·50	2·00	4·50	8·0	12 5	18·0	24 5	32·0	40·5	50 0	60 5	72·0	84·5	98·0	112 5	128·0
	R = 200000	$\frac{1}{b} = \frac{1}{2}$	1·25	5·0	11·25	20·0	31 3	45·0	61·3	80·0	101·2	125 0	151 3	180·0	211·3	245·0	281 3	320·0
		$\frac{1}{b} = \frac{1}{4}$	0·62	2·5	5 62	10·0	15·6	22 5	30·6	40 0	50·6	62 5	75·6	90 0	105·6	122·5	140·6	160·0
	R = 190000	$\frac{1}{b} = \frac{1}{2}$	1·32	5 26	11·8	21·1	32·9	47 4	64 5	84 3	106 5	131 6	159 0	189 0	222·0	257·0	295·0	336 0
		$\frac{1}{b} = \frac{1}{4}$	0·66	2·63	5·9	10 5	16 4	23 7	32 2	42 1	53 2	65 8	79 5	94·5	111·0	128·5	147·5	168·0
	R = 180000	$\frac{1}{b} = \frac{1}{2}$	1 4	5 6	12 5	22 2	34 7	50·0	68·1	88·9	112 5	139·0	168·0	200 0	234 0	275 0	312 0	358 0
		$\frac{1}{b} = \frac{1}{4}$	0 7	2 8	6 3	11 1	17·4	25·0	34·1	44 4	56 3	69·5	84·0	100·0	117 5	137 5	156 0	179·0
	R = 160000	$\frac{1}{b} = \frac{1}{2}$	1 56	6 25	14·05	25·0	39·1	56 25	76 6	100 0	126 5	156·0	199·0	225·0	263·0	306·0	352 0	400·0
		$\frac{1}{b} = \frac{1}{4}$	0·78	3·13	7·03	12·5	19 6	28 1	38 3	50·0	63 25	78·0	94·5	112 5	131·5	153·0	176 0	200·0
	R = 150000	$\frac{1}{b} = \frac{1}{2}$	1·7	6·7	15·0	26·7	41 7	60·0	81·7	106 7	135·0	166·7	202·0	240·0	281 6	326·5	375·0	427·0
		$\frac{1}{b} = \frac{1}{4}$	0·8	3 3	7·5	13·3	20 8	30 0	40 8	53·3	67·5	83 3	101 0	120·0	140·8	163 2	187 5	213 5
	R = 120000	$\frac{1}{b} = \frac{1}{2}$	2·08	8 34	18 76	33 4	52 0	75 0	102 0	133·6	168 8	208·4	252·0	300·0	352·0	408 0	470·0	532·0
		$\frac{1}{b} = \frac{1}{4}$	1·04	4 17	9 38	16 7	26 0	37·5	51·0	66·8	84 4	104 2	126·0	150 0	176 0	204 0	235 0	266 0
	R = 100000	$\frac{1}{b} = \frac{1}{2}$	2·5	10 0	22 5	40·0	62 5	90·0	122 5	160·0	202 5	250 0	302 0	360 0	422·0	490 0	562·0	640·0
		$\frac{1}{b} = \frac{1}{4}$	1 25	5 0	11 25	20 0	31 3	45 0	61 3	80 0	101·25	125·0	151·0	180·0	211 0	245·0	281 0	320·0
	R = 90000	$\frac{1}{b} = \frac{1}{2}$	2·78	11·1	25·0	44 4	69·4	100·0	136 0	177·6	224 0	278·0	336 0	400·0	468·0	544·0	622·0	716 0
		$\frac{1}{b} = \frac{1}{4}$	1 39	5 55	12 5	22 2	34 7	50 0	68 0	88·8	112 0	139 0	168 0	200 0	234 0	272 4	311·0	358·0
	R = 70000	$\frac{1}{b} = \frac{1}{2}$	3 6	14 3	32 0	57·0	89 5	129·0	175 0	229 0	290·0	358 0	432·0	512 0	602 0	700·0	802 0	916·0
		$\frac{1}{b} = \frac{1}{4}$	1·8	7·15	16 0	28·5	44·8	64 5	87 5	114 0	145·0	179·0	216 0	256 0	301 0	350 0	401 0	458 0
	R = 60000	$\frac{1}{b} = \frac{1}{2}$	4 17	16 69	37 5	66 7	104 2	150·0	209·0	266 5	337 5	417 0	503 0	600 0	705·0	818 0	940·0	1064 0
		$\frac{1}{b} = \frac{1}{4}$	2 09	8 34	18 7	33·4	52 1	75·0	102·0	133 3	168 7	208·5	251 5	300 0	352 5	409 0	470 0	532 0
	R = 50000	$\frac{1}{b} = \frac{1}{2}$	5 0	20 0	45 0	80 0	125 0	180·0	245·0	320 0	405 0	500 0	605 0	720·0	845 0	980 0	1125·0	1280·0
		$\frac{1}{b} = \frac{1}{4}$	2 5	10·0	22 5	40 0	62 5	90 0	122·5	160 0	202·5	250 0	302 5	360 0	422 5	490 0	562 5	640·0

beam, for example—with fracture occurring at a critical juncture. Buffers also should be carefully studied on the eighth scale curve layout, as occasions will arise which compel the use of an elliptically-shaped head having the major axis horizontal, the surface of the outer half being curved more sharply in plan to ensure that adjacent buffers on a quick curve shall ride on surfaces and not on the edges.

The matter of speed on the curve is decided by the Permanent Way Engineer's department, and can be related to the super-elevation provided by that department ; the following notes show the connection. A revolving mass is restrained from leaving the truly circular path by the ties which link it with the axis ; a train travelling on a curved track, however, is restrained by the pressure of the wheel flanges against the outer rail. If this rail were not raised above the level of the inner, the effect of centrifugal action, operating through the vertical centre of gravity of the whole machine, would be to overturn the locomotive at comparatively low speeds. By raising, or super-elevating, the outer rail, the vehicle is tilted inwards to the centre of the curve, and an increased arm provided incidentally for the mass of the engine to act in opposition to the overturning effort.

The stability of the unit on any curve is guaranteed so long as the moment of the engine weight acting at the arm Y (see Fig. 6) is at least equal to the overturning force F acting at arm X. But F, the centrifugal force, equals Wv^2/gr (2)

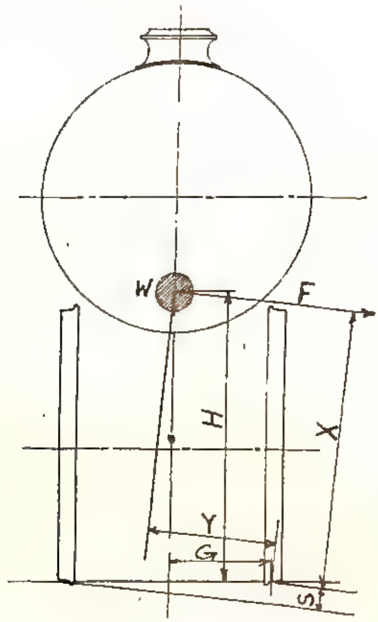


Fig. 6.

where W = weight of locomotive in tons.
v = velocity in feet per second.
g = 32·2
r = radius of curve in feet.

If instead of v we take V as the speed in M.P.H., the formula changes to

$$(5280/3600)^2 \ (WV^2)/gr$$
$$= WV^2/15r \qquad (3)$$

Immediately prior to overturning the forces are equal, *i.e.*, FX = WY, but an increase of F beyond the safe limit, due to too high a speed, will cause a derailment.

Equating, $(WV^2/15r)\ X = WY$ and the safe limit in speed,

$$V = \sqrt{15\ YR/X} \qquad (4)$$

The arm X is approximately equal to H, the height of the engine centre of gravity above rail, less half the superelevation S. Likewise, the arm Y is equal to G + (HS/2G) within reasonable limits.

Reference was made earlier in these notes to the position of the South African system in the world of railways, and the growth of this line during the last 40 years is really remarkable, whether from the point of view of traffic handled, size and power of engine units, weight of trains hauled, etc. Perhaps one of the most striking testimonies to their progress was that given by the then General Manager, Mr. W. W. Hoy, in his annual report of 1916, the position being such, even in those days, that he could compare it favourably, not with other systems of similar gauge, but with the 4′-8½″ gauge lines in the mother country. During the 25 years since his comparison was made, the advance has been greater than even Mr. Hoy could have anticipated, powerful passenger and freight locomotives, besides heavy articulated engines of the 4-8-2 — 2-8-4 class, being in service on different sections of the system, while the coaching stock includes sleepers and dining cars equal in every respect to any in the world. An extract from Mr. Hoy's report is given below, and forms an interesting commentary on the possibilities of the 3′-6″ gauge.

"Experiments carried out in South Africa show—and the experience of other railways confirms this—that speeds ranging to 50 and even 60 M.P.H. can, with safety and comfort, be accomplished on the 3′-6″ gauge track when maintained at a high standard and free from sharp curves. . . .The main lines are gradually being improved by heavier metals, grade easement and curve flattening, and the road bed generally is being brought up to a standard which will admit of safe and comfortable travelling at high speeds. By steady development in all directions, I see no reason why all reasonable demands for rapid transit, heavy loads and economical working should not be met by South Africa's standard gauge."

The following is a comparison of recently built engines for the British 4′-8½″ lines and the 3′-6″ African system, excluding such types as the Mallet and Garratt locomotive, of which, however, the South African Railway boasts some fine engines.

	S. African Rly.	*L.M. & S. Rly.*
Type, ...	4-6-2 Tender	4-6-2 Tender
Max. Tractive Effort—lbs. (at 85% M.E.P.)	40,315	40,000
Max. Adhesive Load—tons	59·5	67·1
Max. Engine Weight (less tender)	98·9	108·1
Coupled Wheel Diameter—ins.	72	81
Cyl. sizes—bore × stroke—ins.	24 × 28 (two)	16½ × 28 (four)
Total Heating Surface—sq. ft.	3534	3663
Grate Area—sq. ft. ...	62	50

Climate.—This must be taken into account from several points of view, a moist climate naturally affecting the rail surface and the

adhesion obtainable. In some districts where the track passes through deep cuttings, a serious loss of power is sustained at certain periods of the year due to the heavy fall of vegetation on the track, and allowance must be made for these and similar contingencies.

In countries where extremely low temperatures prevail, pipes and tanks for oil and water must be adequately protected from the effects of frost, and drain cocks fitted at vital points. On some systems the presence of snow on the track has compelled the fitting of strong wire brushes at each end of the locomotive together with "pusher" bars, the bars clearing the bulk of the newly-fallen snow to the side of the track, and the brushes virtually scrubbing the remaining snow from the rail face, an application of sand where necessary restoring the adhesion to the equivalent of a dry rail, or nearly so.

Where sand-storms are frequent, guards or casings are fitted to protect the motion and running gear, as well as the engine crew in the cab; similarly, in tropical countries, adequate provision must be made to protect the personnel from the heat of the sun's rays, to ventilate the cab and, as far as possible, maintain a cool roof by the circulation of air between its upper and lower sections. Heavy rains must be guarded against, as sandbox lid-covers, which may be perfectly tight in a temperate zone, will sometimes admit water under conditions of tropical severity.

Class of Service.—Although small lines will use the same engine on a variety of services, the larger systems with their heavier traffic can afford to employ special types for particular services. Briefly, these may be classified under three heads—passenger, goods and shunting engines. These in turn are further subdivided under local and long-distance express passenger engines; mixed traffic; heavy goods and mineral locomotives; with, finally, light and heavy shunting engines.

Local passenger services may be operated by any one of several classes—often a tank engine may be delegated to the job. The requirements, from a designer's viewpoint, usually call for moderate cylinder and boiler power, with wheels for top speeds of approximately 50 to 60 miles per hour, but where the traffic is of a suburban character, with peak loads and frequent services at rush hours, larger cylinders, smaller wheels and increased boiler power become necessary.

For long-distance express services, tank engines are occasionally employed, particularly the 4-6-4 class, with driving wheels around 6′-0″ diameter, but the tender engine is more frequently commissioned for long non-stop runs owing to the greater reserves of water and fuel which a tender will carry. A larger wheel (6′-0″ to 7′-0″) is essential for high speed and possibly three, or even four, coupled axles to obtain adhesion commensurate with the power

demand. If the engines are to operate at speed over track and bridges which are not up to the maximum strength on the railway in question, the adoption of a three- or four-cylinder design will be considered in order to limit the hammer-blow. Cylinder and boiler power must be safe for the worst conditions, usually observed at speed, and the cylinder dimensions should be planned to enable the locomotive to start a long train from rest with no outside assistance at any point on the section to be worked.

Locomotives formerly known as goods engines now work such a variety of services that their field has become very diverse, and where capital is available to permit the employment of special engines for particular duties, the one-time goods engine may be to-day a fast goods (or mixed traffic) locomotive, a heavy goods, or a mineral engine, the chief notable difference being found in the wheelbase and the size of the driving wheels. The first of the three types referred to hauls the Blue Arrow and similar trains on a fast schedule, and will be of the 2-6-0 or 2-6-2 type ; the heavy goods will probably be a 2-8-0 or 2-8-2 class, and the mineral, a heavy slow-running machine of the eight or ten coupled locomotive class, probably with neither bogie nor truck.

In the shunting locomotive sphere, the unit may have light or heavy duties to perform, and should be designed with one object in view—the exertion of the maximum tractive effort consistent with its adhesion. A light shunter would be in the region of 20 to 30 tons in weight fully loaded, and have two or three coupled axles carrying all the weight. A heavy machine would be from 50 to 55 tons in weight, and have three, or possibly four, coupled axles sustaining the entire engine. Cylinder dimensions and boiler power in all shunting locomotives are related purely to tractive force, though in rare cases such an engine may have a moderately long stretch of haulage from the marshalling yard to the docks, in which case a measure of speed would be called for and the wheel diameter would be increased slightly. To assist in traversing all yard curves, shunting locomotives should have the minimum wheelbase possible.

Fuel.—The fuel to be used will be indicated by the client in his enquiry specification, and will depend naturally on local supplies, if any. Many fuels are in use in different parts of the world, and since the early days when coke was consumed so largely, coal, oil, wood and even straw have been employed. In some tropical countries, sugar canes have been used (naturally after the juice has been extracted) and to-day reports mention that coffee, when oil-treated, has given excellent results.

Coal, however, is probably the most commonly used, with oil a close runner-up. High evaporation can be achieved with good qualities of either fuel, and the firebox design is not greatly different

for these, although a long box is preferable for oil firing and a more or less square grate for coal. The inner box and the firepan must be lined suitably with firebrick when oil is employed, and the firepan then becomes virtually a sealing chamber beneath the firebox, with a suitable arrangement of air inlet passages and dampers.

Up to recent years the grate width was fixed by the maximum obtainable between the main engine frames, but this limitation no longer exists in designs which employ a rear carrying unit, and the grate can then be to any width consistent with the load gauge. Thirty years ago a large grate usually meant a long one, the firebox in most cases being arranged between the frames, and the fireman's energy (with coal firing) could be severely taxed on a long run with a heavy train. The modern grate, more or less square in plan, gives a much better steam raiser with coal fuel for appreciably less output of manpower.

Wood fuel calls for a long narrow grate, and the same design would apply to sugar canes or similar fuels.

For oil fuel a steel inner firebox should always be used, as the action of the oil is most destructive on the copper normally employed in these structures.

The designer, then, must have full particulars regarding the fuel his engine is to burn, its calorific value in B.T.U. and the distance between supply bases in order to determine whether a tank engine will meet the case, or, alternatively, the size of tender required.

Water.—For similar reasons, information is required as to the character of the water available and the points where tanks may be refilled. If the water is known to be strong in acids, a sample should be obtained and analysed, as special precautions will be necessary in the design of the boiler and decisions taken regarding the materials to employ for the inner firebox and tubes. Special wash-out facilities may be required, and it may be desirable to provide such equipment as feed water heaters with top feed, or special boiler cleaning devices. For certain sections having bad water, the Indian State Railways have boilers specially treated on the inside surfaces of the lower half of the barrel, and on many lines with water having a strong acid content, special devices are included in the boiler design to intensify the circulation and avoid any risk of steam pockets forming at dangerous points.

The quality of the water to be used may also be a very serious factor in the life of the inner firebox and the tubes, and every possible precaution should be taken to assist the client in the maintenance of his boiler and the securing of a satisfactory life for the parts referred to, since their renewal can be a very costly affair.

To facilitate the running of non-stop long distance trains on British main line railways, water troughs are provided at suitable

points, these being kept filled to the correct level by an automatic device. The tender has a mechanically-operated scoop which is lowered by the fireman as and when required, the momentum of the train causing the water to flow up the scoop pipe into the main tank. These devices are useful, but in frosty weather the spray, when frozen, has been known to give considerable trouble, and this fact needs to be borne in mind when designing the scoop and surrounding equipment.

Type of Engine Selected.—Where the particular type of engine to be used is left to the tenderer to put forward, the information already to hand giving the precise duties the engine will be required to carry out affords a valuable groundwork for a designer to proceed on. He can now decide whether a tank or tender engine is essential, or, alternatively, whether an articulated unit will give better results ; what coupled wheelbase will be required, the sizes of driving wheels, cylinders and boiler proportions. The latter will settle whether a bogie, or truck—or both—must be employed, this depending on whether the overhang of boiler, etc., at front and rear ends is excessive.

Where the power of an engine demands an appreciably heavier axle loading than is permissible for the type of unit which meets the general requirements, consideration must be given to the desirability of employing an articulated engine of the Garratt or Mallet type. This involves higher capital cost, but it has been found the only feasible way of combining a low, or relatively low, axle load with high overall tractive power. On the other hand, if the extra power demand is for short periods only, the solution may be the fitting of a booster engine to the trailing truck, or bogie, the auxiliary engine cutting in at low speeds and assisting through a difficult phase without stalling the main engine.

Thus the classification of the new engine can be finally fixed, and in view of the several methods chosen for designating the numerous types, Table 3 is included to give the various symbols, etc., for any wheel layout.

The following table does not include the Garratt engine, as this is usually merely a duplication of the ordinary locomotive with the rear ends to the centre. Thus a 4-8-2—2-8-4 Garratt would be a 2D1-1D2 by the German method of reckoning.

Most locomotives to-day, certainly in Britain and the U.S.A., are simple engines, but on one or two South American lines and on the Continent, two- and four-cylinder compounds are employed. A few of the Smith three-cylinder compounds are still doing useful service in Britain, but this type is not being repeated. The Central Argentine Railway has a number of heavy two-cylinder compounds in service, and the Bengal Nagpur Railway is operating several 4-6-2 type four-cylinder compounds, built on the de Glehn system,

TABLE 3.

WHEEL ARRANGMT.	NOTATION BRITISH & USA	FRENCH	GERMAN	TYPE
OO	0-4-0	0 2 0	B	
oOO	2-4-0	1 2 0	1 B	
OOo	0-4-2	0 2 1	B 1	
oOOo	2-4-2	1 2 1	1 B 1	COLUMBIA
ooOO	4-4-0	2 2 0	2 B	AMERICAN
OOoo	0-4-4	0 2 2	B 2	FORNEY [illegible]
oOOoo	2-4-4	1 2 2	1 B 2	
ooOOoo	4-4-4	2 2 2	2 B 2	READING
ooOOo	4-4-2	2 2 1	2 B 1	ATLANTIC
OOO	0-6-0	0 3 0	C	
oOOO	2-6-0	1 3 0	1 C	MOGUL
oOOOo	2-6-2	1 3 1	1 C 1	PRAIRIE
OOOo	0-6-2	0 3 1	C 1	
ooOOO	4-6-0	2 3 0	2 C	
OOOoo	0-6-4	0 3 2	C 2	FORNEY SIX COUPLED
ooOOOoo	4-6-4	2 3 2	2 C 2	BALTIC
ooOOOo	4-6-2	2 3 1	2 C 1	PACIFIC
oOOOoo	2-6-4	1 3 2	1 C 2	ADRIATIC
OOOO	0-8-0	0 4 0	D	
oOOOO	2-8-0	1 4 0	1 D	CONSOLIDATION
OOOOo	0-8-2	0 4 1	D 1	
oOOOOo	2-8-2	1 4 1	1 D 1	MIKADO
ooOOOO	4-8-0	2 4 0	2 D	
OOOOoo	0-8-4	0 4 2	D 2	
ooOOOOo	4-8-2	2 4 1	2 D 1	MOUNTAIN
oOOOOoo	2-8-4	1 4 2	1 D 2	BERKSHIRE
ooOOOOoo	4-8-4	2 4 2	2 D 2	CONFEDERATION
OOOOO	0-10-0	0 5 0	E	
oOOOOO	2-10-0	1 5 0	1 E	DECAPOD
OOOOOo	0-10-2	0 5 1	E 1	
oOOOOOo	2-10-2	1 5 1	1 E 1	SANTA FÉ
ooOOOOOo	4-10-2	2 5 1	2 E 1	OVERLAND
ooOOOOO	4-10-0	2 5 0	2 E	MASTODON
oOOOOOoo	2-10-4	1 5 2	1 E 2	TEXAS

the principle having proved satisfactory on the B.N.R. many years ago, when 4-4-2 type express engines were built on the same system and fully tested. The chief attraction of any compound locomotive is its higher thermal efficiency, and when of the four-cylinder type there are the added advantages of improved balancing and lower stresses on the vital parts. Against this, however, must be set the higher costs of construction and maintenance, and it becomes essentially a C.M.E.'s responsibility to decide whether the new engine shall be simple or compound, since he alone has the means of estimating the issues on which the directors will form their judgment.

Superheating, on the other hand, is by now almost universally

adopted for all main line engines, those forming the non-superheating group of new locomotives belonging to the shunting and frequent stopping services, as well as those operated by works contractors, steelworks, collieries, etc., where the extra cost of the equipment is not justified by economies in operation.

Train Resistance.—Coming now to basic calculations upon which the design must stand, the first to be made will obviously be those which show the work to be done. Resistances are five-fold in origin—rolling, grade, curve, wind and acceleration—and, being considered in the order stated, will be determined in terms of pounds per ton for vehicles of like type in the train.

Rolling Resistance is that which is produced when a vehicle is started from rest or moved at any speed along a dead level track. The amount varies according to the speed, and Figs. 7, 8 and 9 show curves used by the Baldwin Locomotive Co. for locomotives, passenger and freight cars respectively, from which it will be seen that the starting friction is approximately 16 lbs. per ton of static loading at the rails, falling to around 5 or 6 lbs. per ton at 10 M.P.H.

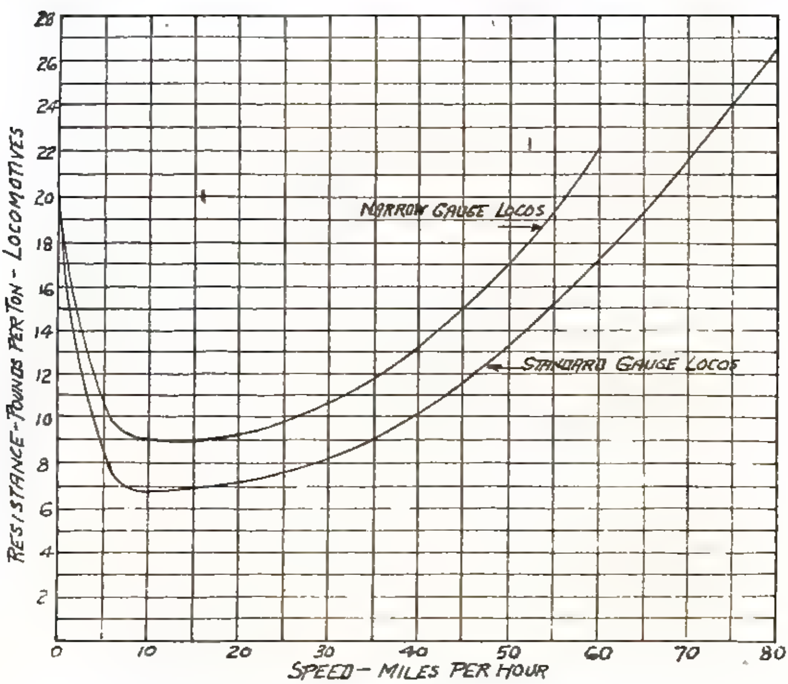


Fig. 7.

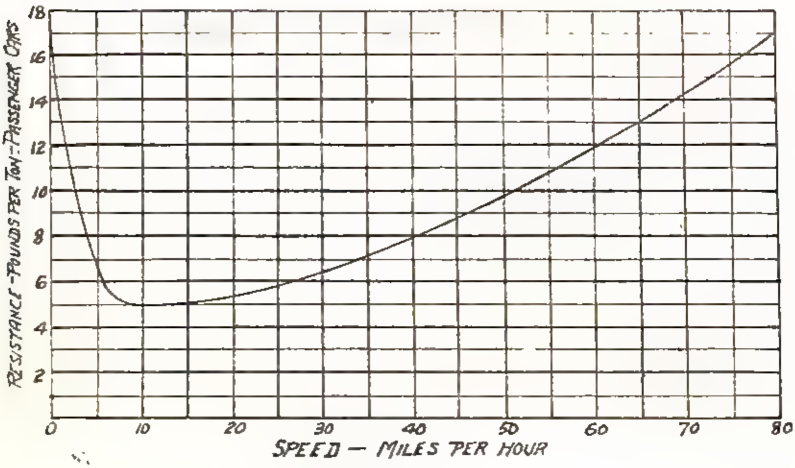

RESISTANCE - POUNDS PER TON: PASSENGER CARS
18
16
14
12
10
8
6
4
2
0
10
20
30
40
50
60
70
80
SPEED — MILES PER HOUR

Fig. 8.

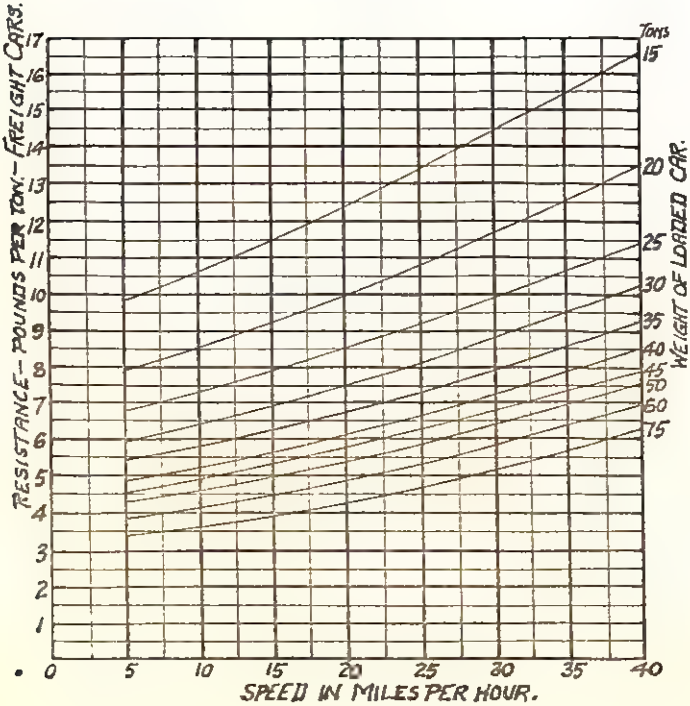

RESISTANCE - POUNDS PER TON: - FREIGHT CARS.
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
0
5
10
15
20
25
30
35
40
SPEED IN MILES PER HOUR.
TONS
15
20
25
30
35
40
45
50
60
75
WEIGHT OF LOADED CAR.

Fig. 9.

and rising thereafter according to the speed. The following formulae are the bases for these graphs :—

Standard gauge locomotives,	$R_r = 4{\cdot}84 + 0{\cdot}0034\,V^2$	(5)
Narrow gauge locomotives and those for light standard gauges,	$R_r = 5{\cdot}6 + 0{\cdot}0045\,V^2$	(6)
Passenger coaching stock,	$R_r = 4{\cdot}84 + 0{\cdot}0019\,V^2$	(7)

Freight cars, $$R_r = 1{\cdot}687 + \frac{9\,(106 + 2V)}{8\,(W + 1)} + 0{\cdot}0011V^2 \quad (8)$$

V being the speed in M.P.H. ; R_r the rolling resistance in lbs. per ton, and W the weight of cars in tons. (*Note*—1 ton = 2240 lbs.).

In the case of passenger vehicles, as all new stock is of the double-bogie type, the curve given will be suitable for general all-round service, but in the case of freight cars and wagon stock, as the first are usually of the double-bogie type, but much more heavily loaded than any passenger car bogies, the curve shown would seem to give lower values than would be expected for four-wheel wagons so much in use in Great Britain, and some allowance for this must be made.

Resistances for locomotives must include that necessary to overcome the internal friction of the power unit, as well as that for the vehicle—hence the higher values per ton. These greater resistance figures, however, must only be used for the coupled wheel loads, those for the bogie, truck and tender axles being regarded as part of the train weight and included in that group.

The graphs and formulae given can be compared by the reader with other values as suggested by different authorities, some of which are given below.

Mr. Lawford H. Fry's formulae are for

Locomotives.

4 wheels coupled,	$R_r = 8{\cdot}5 + {\cdot}0974\,V + {\cdot}004\,V^2$	(9)
6 wheels coupled,	$R_r = 10{\cdot}08 + {\cdot}126\,V + {\cdot}004\,V^2$	(10)
8 wheels coupled,	$R_r = 13{\cdot}34 + {\cdot}48\,V + {\cdot}004\,V^2$	(11)

Coaching and High Capacity Wagon Stock.

$$R_r = 3{\cdot}6 + {\cdot}03\,V + {\cdot}0027\,V^2 \quad (12)$$

Goods Stock (4 or 6 wheels).

$$R_r = 3{\cdot}6 + {\cdot}03\,V + {\cdot}0022\,V^2 \quad (13)$$

In conducting some recent experiments for the L.M.S. Railway, in connection with the air resistance of trains, Mr. Johansen proposed a rolling resistance of

$$R_r = 4 + 0{\cdot}025\,V + 0{\cdot}00166\,V^2 \quad (14)$$

Deeley gives $R = 3{\cdot}25 + (V^2/281)$ (15)

Laboriette recommends $R = 3 + (V^2/290)$ (16)

(each of the last two values are only advised for speeds exceeding 20 M.P.H.), and

Wolff proposes $$R = 3\frac{(V+12)}{(V+3)} + \frac{V^2}{300} \qquad (17)$$

Other formulae are in use, such as Aspinall's, Wellington's, Strahl's (the last generally employed for German and continental stock) and others, and this subject has intrigued engineers to such an extent that each investigator seems to produce a separate value differing little, in the main, from those already quoted.

Grade Resistance.—When a train moves up an incline, the resistances to be overcome comprise that due to rolling plus that involved in raising the train through a given height against the effect of gravity. Thus if the grade be 1 in 400, the work due to gravity is that done in raising the train through one foot for every four hundred travelled, and is taken as 2240/400 pounds per ton. Taking the grade as 1 in n feet, the resistance

$$Rg = 2240/n \text{ lbs. per ton} \qquad (18)$$

and Fig. 10 gives the graph for this value.

Sometimes the inclination is stated as x per cent.; in such case the resistance $Rg = 2240\,x/100$.

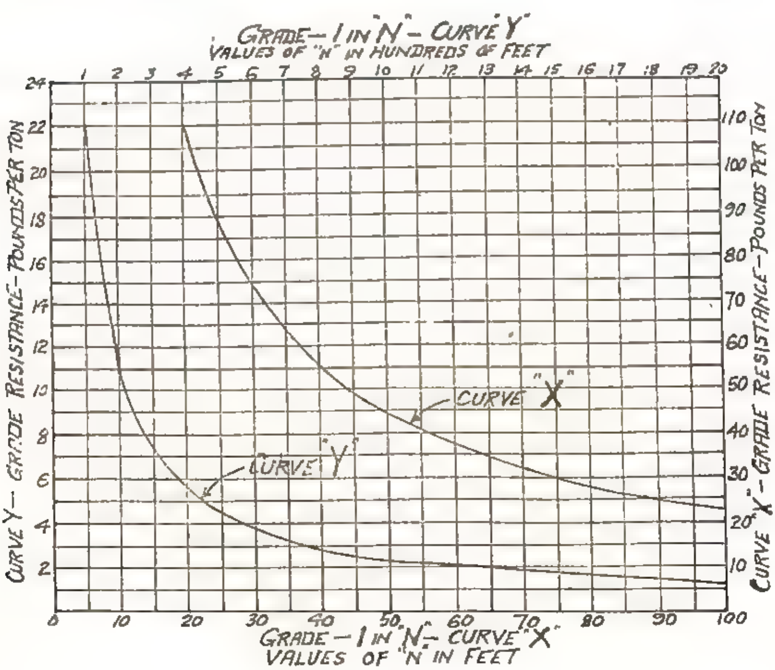


Fig. 10.

Curve Resistance.—The value taken here by American builders is X lbs. per ton per degree of curvature, X representing, for the Baldwin Locomotive Co., 0·70 to 1·00 lb. per ton, and for the American Locomotive Co. 0·80 lbs. per ton (of 2000 lbs.). The first value is for large and small capacity cars, the latter having more axles per train of a given load, and therefore imposing a greater tax on the engine while curving. For eight coupled engines, however, the figure of 1·5 tons should be taken, with a slight increase for ten or twelve coupled locomotives.

The measure of the curve in degrees is found by taking a chord of 100 feet and a radius of 5730 feet, the angle enclosed by two radials being 1 degree. If the curve is specified as being of 573 feet radius, the number of degrees enclosed by the 100 feet chord will be approximately 5730/573, or ten. Likewise, the radius in feet can be found by dividing 5730 by the number of degrees specified, and although this abbreviated method is not strictly accurate, the error is negligible. The 100 feet given as the chord is actually measured on the rail (the arc) when checking a curved track, but the error here also is quite small.

An alternative method of computing curve resistance is by Morrison's rule, and if this be employed the result will be the sum of the resistances as worked out for each type of vehicle in the train; not difficult where passenger or freight cars of double bogie design are employed, but otherwise rather tedious, and frequently impossible to determine. The rule, however, is

$$R_c = 2240\,F\,(G+L)/2r \tag{19}$$

where r = radius of curve (in feet).
L = length of rigid wheelbase (in feet).
G = gauge of railway.
F = coefficient of friction of wheels on rail, varying from 0·1 to 0·27, according to weather conditions.

Where a curve falls on a grade, the latter is frequently compensated locally to allow for the effect of the curve on the tractive power available. A resistance of 0·7 to 1·00 lb. per degree of curvature is equivalent to that involved by a grade of 0·04 per cent. and the compensation referred to is effected by a reduction of grade such that the resistance gained equals the extra necessary for the curve.

Air or Wind Resistance.—This may take two forms—head winds directly opposing the forward progress of the train, and side winds having a similar result, but achieved by a flank pressure which forces the vehicles across the track, setting up increased flange friction.

No allowance can be made for side winds, as these are usually not sufficiently constant and therefore outside the scope of useful

calculations. For the effect of head winds, the formula used by the A.L. Co. is

$$\cdot 002\ V^2A \tag{20}$$

where V is the speed in M.P.H. and A represents the maximum head-on area of the engine and train. It should be noted that for a train travelling at 60 M.P.H. against a head wind of 40 M.P.H. V must be made equal to the sum of the two velocities combined. The value of A usually taken is from 100 to 120 square feet, and if we assume the higher value for the speeds just quoted, the tractive effort of the engine must cover for the total air resistance of $\cdot 002\ (60+40)^2 \times 120 = 2400$ lbs.

Mr. Dendy Marshall gives a formula for air resistance calculations of $0{\cdot}00228\ V^2$, in which V is the speed (M.P.H.) and the result is in pounds per square foot of end surface. Thus, in the example already quoted, D.M.'s formula would give

$$0{\cdot}00228 \times 100^2 \times 120 = 2736 \text{ lbs.}$$

As this resistance varies directly as the square of the speed, it is obviously necessary to include it when really high speeds (say over 60 M.P.H.) are contemplated, particularly so if the line crosses open country where head winds are frequently encountered. Under such circumstances the C.M.E. may with ample justification adopt streamlining in order to reduce the size of his power unit.

In this connection it is interesting to note the results of experiments made by the L.M.S. Railway on scale models of express engines and trains—with and without streamlining—in wind tunnels, and described by Mr. F. C. Johansen in the paper he read before the Institution of Mechanical Engineers in November, 1936. The air resistance of a train of conventional British design, according to the above writer, is equivalent to about $0{\cdot}0016V^2$ pounds per ton of train weight, where V is the speed in still air in M.P.H., and represents upwards of half the total train resistance on the level track at speeds over 80 M.P.H.

The above formula, like that already quoted in this section, has three components; the constants in the two differ only in small degree, and the speed term is common to both. But where results by the first formula are dependent on the total end area of the train measured in square feet, those by the L.M.S. Rly. formula are related to the train weight. Fig. 11 graphs the results by the two formulae for different speeds, assuming a train of 600 tons weight with an end area of 120 sq. feet.

Resistance due to Acceleration.—The engine power required to overcome the resistances already tabled—rolling, grade, curve and air—will enable definite speeds to be attained at different points on the system, but whenever a speed has to be increased, the resistance of the train to the increased motion must be taken into account, along with that of the wheels to the corresponding increased

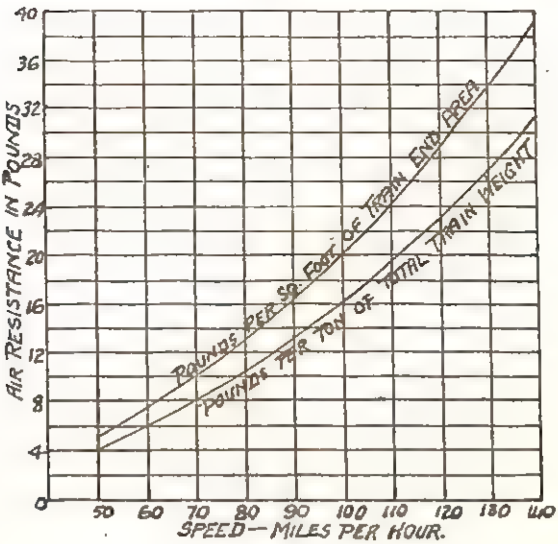


Fig. 11.

speed of rotation. Both are covered in the formulae generally employed, which give

$$Ra = 70 \quad (V^2 - v^2)/S \tag{21}$$

and

$$Ra = 95{\cdot}6 \ (V - v)/t \tag{22}$$

where Ra = the accelerating force in pounds per ton.
S = distance in feet through which Ra acts.
t = time in seconds during which Ra acts.
V = higher velocity in miles per hour, and
v = lower velocity in miles per hour.

When making up the total of engine and train resistance the figure taken for the locomotive should be applied to the adhesive weight only for rolling resistances as previously stated, the bogie, truck and tender loads being detailed with those for the train. The values found for the grade and also for acceleration apply equally to locomotive and train, likewise that for the curve, unless the power unit is an eight, ten or twelve coupled engine, when the adhesive weight alone would be multiplied by the higher value suggested, and the remaining loads—engine and train—by the lower figure selected. Air or wind resistance will be related to the total engine and train weight, or to the end area opposing movement, as preferred.

Tractive Effort.—The power of any locomotive to start and haul a train is measured in pounds at the rail, and is limited by the load imposed at the line of contact between rail and driving wheels,

otherwise known as the adhesive, or gripping, load. When the latter falls below the value necessary for the power applied, slipping of the wheels ensues.

The measure of the tractive effort of any engine, for starting, is determined solely by the size of the driving wheels, the cylinder dimensions and the mean effective steam pressure in the cylinder. Assuming D to represent the diameter of driving wheels, d that of the cylinder bore, L the length of the stroke (all in inches) and P the mean effective steam pressure in pounds per square inch.

TE, the tractive effort

$= d^2LP/D$	for 2 cyl. simple locos.	(23)
$= 1{\cdot}5d^2LP/D$	,, 3 ,, ,, ,,	(24)
$= 2d^2LP/D$	,, 4 ,, ,, ,,	(25)

The value P is generally taken at 85 per cent. of the maximum boiler steam pressure, and is a safe figure to employ, providing the maximum cut-off is made not less than 75% of the stroke.

For two-cylinder compound locomotives the A.L. Co. give

TE, when working compound	$= D^2SPC/2W$	(26)
when working simple	$= {\cdot}85d^2SP/W$	
	$= 26 \times (1{\cdot}7/CR)$	
	$= 1{\cdot}3D^2\,SPC/2W$	(27)

For four-cylinder compound locomotives.

TE, when working compound	$= D^2SPC/W$	(28)
when working simple	$= (2 \times {\cdot}85)\, d^2SP/W$	
	$= 28 \times (1{\cdot}7/CR)$	
	$= 1{\cdot}3\, D^2SPC/W$	(29)

The terms employed in these formulae are as follows :—

- d = diameter of H.P. cylinder in inches.
- D = diameter of L.P. cylinder in inches.
- S = length of stroke in inches.
- P = boiler pressure in pounds per sq. inch.
- C = constant, taken as ·52 (see Table 4).
- W = diameter of driving wheels in inches.
- R = ratio of L.P. to H.P. cylinder volume and taken as 2·5 (see also table 4).

These American builders point out that results by formulae 27 and 29 give the tractive effort when just moving. At slow speeds the T.E. when working simple will exceed the power when working compound by approximately 20 per cent.

They further state that on superheater compound engines, in order to divide the work equally between the high and low pressure cylinders, the cut-off in the L.P. cylinder should be approximately 5% later than the cut-off in the H.P. cylinder for a cylinder ratio

of 2·5 to 1. This difference in cut-off should be reduced for higher cylinder ratios to zero for a ratio of 2·75 to 1, and increased for lower cylinder ratios to approximately 10% for a ratio of 2·2 to 1.

TABLE 4.

Per Cent. Cut-Off H.P. Cylinder.	Ratio of L.P. to H.P. Cylinder Volume.						
	2·2	2·3	2·4	2·5	2·6	2·7	2·8
90			·571	·557	·542	·528	·513
89			·565	·550	·536	·521	·507
88		·573	·559	·543	·529	·515	·500
87		·567	·552	·537	·523	·509	·494
86	·575	·560	·546	·531	·517	·502	·489
85	·570	·555	·540	·526	·511	·497	·483
84	·564	·550	·534	·520	·506	·491	
83	·559	·544	·529	·515	·500	·486	
82	·553	·541	·524	·510	·496		
81	·548	·534	·520	·505	·490		
80	·543	·531	·515	·500	·486		

The above values of tractive effort hold within certain limits of speed, but when these are exceeded the boiler is unable to meet the demand for steam at the cut-off in operation, and this must be reduced. As speed continues to increase, the cut-off is further restricted, with a consequent fall in tractive power at each change in the cut-off. As a result, the actual calculations for the T.E. become rather difficult to make when the speed is continually subject to change.

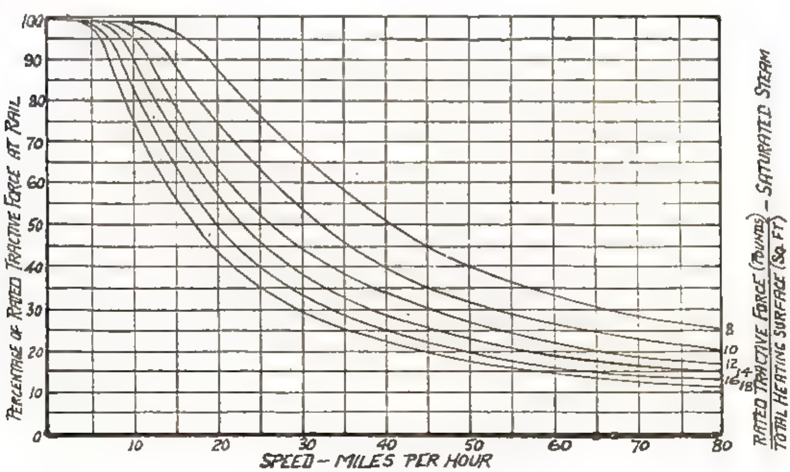


Fig. 12.

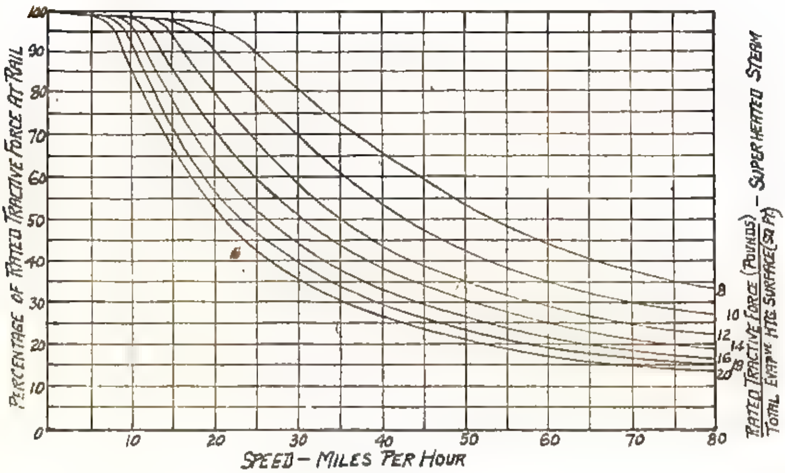


Fig. 13.

The Baldwin Locomotive Co. give two graphs in their pocket book that meet the case just referred to, these being shown in Figs. 12 and 13. First, determine the ratio between the rated tractive effort and the total heating surface ; if the engine is superheated, the ratio must be between the tractive effort and the evaporative heating surface. Having found this value, use the curve nearest to that ratio for the engine under review, and ascertain the percentage value for the speed in question. The actual tractive effort at that speed will be the maximum static tractive effort times the percentage value found.

Where a compound engine is in process of design, the question of cylinder ratios will arise and the diameter of the high and low pressure cylinders must be settled to ensure that under average conditions the work done in any one cylinder shall be equal to that done in each of its fellows. This will be obvious in order to split up the total work done equally over the several cylinders provided, and also to equalise the stresses ; on two-cylinder compounds this is particularly essential, as unequal work in the two cylinders would set up a dangerous twisting of the engine athwart the track. Naturally the work cannot be equal at all cut-offs in the two cylinders—H. and L.P.—and the aim should be to effect equality for the cut-off most frequently to be employed, as far as can be estimated. The difficulty can be overcome to some little extent by separately notching up the H. and L.P. valves, but even with this refinement the ideal to be aimed at is only partially possible of achievement.

The most desirable ratio can be arrived at by the C.M.E.'s staff from the data obtained from the running of earlier compound engines on their system, and this value should be covered by the specification. Mr. J. G. Gairns, in his book on "Locomotive Compounding and Superheating," gives the ratio for two-cylinder compounds as usually about 1 to 2 or 1 to 2·25, though occasionally 1 to 2·5 and 1 to 2·75, or even 1 to 3, are employed. For three-cylinder compounds, where only one H.P. cylinder is to be used, the three cylinders will be about equal in diameter, or the L.P. cylinders may be slightly larger, giving a proportion of 1 to 2, or 1 to 2·25. He quotes the famous Smith compounds on the old Midland Railway, where a single 19″ dia. H.P. cylinder and two 21″ dia. L.P. cylinders are fitted. On Webb three-cylinder compounds with one L.P. and two H.P. cylinders, the diameters have been 15″ (or 16″) and 30″, giving a ratio of 1 to 2 or slightly less.

Four-cylinder compounds have a varied scale of ratios, running from 1 to 1·7 up to 1 to 3, but where the lower value has been used, the influencing factor has been the proportioning of the H.P. cylinders sufficiently large for them to start the train unaided. Although it is desirable to spread the work to be done evenly over all cylinders in a four-cylinder compound, it is not so essential as where two cylinders only are employed, for reasons that will probably be self-evident.

The piston strokes are usually equal for H. and L.P. cylinders, but if any difference is to be made, the ratios will be computed then on the cylinder volumes and not merely on the basis of piston diameters.

Drawbar Pull.—This can only be arrived at correctly by actual test on a dynamometer car attached immediately behind the engine, but a theoretical figure can be calculated by deducting the power absorbed in hauling the engine (and tender if any) from the calculated tractive effort.

Horsepower.—This is the product of the tractive effort (in lbs.) and the speed (in M.P.H.) and is obtained from the formula

$$\text{H.P.} = (\text{TE} \times \text{Speed})/375 \qquad (30)$$

The result is often referred to as the theoretical rail horse-power, that taken at the drawbar being named the drawbar horse-power, and obtained by substituting the drawbar pull for the tractive effort in the above formula.

Adhesion.—Reference has already been made to the necessity for providing sufficient weight at the rail to ensure the effective use of the power transmitted through the coupled wheels. For the determination of preliminary resistance values a certain type of engine can usually be assumed, with reasonable accuracy, but after all calculations are made for the combined resistances of

engine and train under extreme conditions, the tractive effort can be settled and the adhesion required arrived at, using the factor compatible with climatic and other conditions. The adhesive load divided by the maximum axle load will decide the number of coupled axles required. Should the tractive effort be 30,000 lbs. and the factor of adhesion 0·2, the load on the rail at the coupled axles will be 30,000/0·2 or 150,000 lbs. With a maximum permissible axle load of 18 tons (40,320 lbs.) the number of coupled axles necessary would be four, and the engine an eight-coupled, with or without bogie and truck, the necessity for these being determined by the boiler power and other factors.

Coupled Wheel Diameter.—This dimension, one of the factors in the tractive effort calculations, is primarily fixed with almost entire regard to the matter of speed when the unit is to be an express engine. Otherwise, the wheel diameter must fit in with the general scheme as regards power and speed. It will also be influenced to some extent by the Railway Company's standards and tyres kept in stock, if these are within reasonable dimensions of the size found necessary by calculation. The following table gives an idea as to the wheel sizes in most general use.

Gauge of Railway.	*Class of Engine.*	*Coupled Wheel Diam.*
5′-6″ to 4′-8½″	Express	6′-3″ to 7′-0″
	Mixed Traffic	5′-3″ to 6′-0″
	Goods	4′-6″ to 5′-3″
	Mineral	4′-0″ to 4′-6″
	Shunting	3′-6″ to 4′-0″
3′-6″	Passenger	4′-6″ to 6′-0″
	Mixed Traffic	4′-0″ to 4′-6″
	Goods	3′-3″ to 4′-0″
Metre	Passenger	4′-9″
	Goods	3′-0″ to 4′-0″

Cylinders.—These also must be determined with reference to the required power, and the question of duplication with existing units considered. This may not be possible in its entirety, but where cylinders of similar size are in operation component parts from these—pistons, valves, covers, etc.—may be worked in.

The stroke offers some latitude for change where an advantage is desired, and with large-coupled wheels a longer stroke is possible. In British practice, 26″ is usually the maximum employed, but the G.W. Railway has several classes using 28″ and even 30″ strokes.

To facilitate the determination of cylinder and coupled wheel proportions, a slight modification to the tractive effort formula for simple engines may be helpful. The customary ratios of diameter and stroke will be found to average 2 to 3, and if we assume *d* and L in formula 23 to have this ratio, the formula becomes

$$(d^2 \times 1{\cdot}5d \times \mathrm{P})/\mathrm{D} = 1{\cdot}5d^3\mathrm{P}/\mathrm{D} \qquad (31)$$

from which d and L can be found, and any adjustment made that may be deemed necessary.

The position of the cylinders, whether to be inside the frames or outside, may be decided for the designer by the client's specification, but if not, consideration of the following "pros" and "cons" will assist towards a settlement. Taking the case of inside cylinders, the following are obtained by their employment :—

1.—A minimum of oscillation, or transverse movement athwart the track. When we have an engine with a fairly long wheelbase, however, this advantage is of very minor value.

2—Less condensation of steam owing to the sheltered position. Then, in many engines having inside cylinders, they form the bottom of the smokebox ; hence the heat from the furnace gases assists in maintaining a somewhat higher temperature than that of cylinders placed outside the frames.

The disadvantages of inside cylinders are—

1—The crank axle is a source of great weakness and is, unless properly and carefully designed, always liable to breakdown.

2—It is, further, a very expensive item to manufacture, and entails heavy expenditure for one item.

3—The cylinder diameter is limited owing to the width between frames, 19″ dia. in this country being the maximum possible on the standard gauge, except with some extraordinary method of design. This size is also limited by the centres of crank journals, as we require, besides these two journals, the remaining two for the axle-boxes, the four crank cheeks and a length of straight axle for mounting the eccentrics. To provide for these in a space of 4′-5½″ or thereabouts requires no small amount of skill when large cylinders are to be employed.

4—Motion and parts less accessible with inside cylinders for lubrication and adjustments.

5—Balance weights in the wheels require to be much larger for inside cylinders.

The advantages and otherwise for outside cylinders are practically per contra to the above, the cost being considerably reduced, taken all round.

The number of cylinders which are to be used may also be fixed for the designer, but where he has a free hand, considerations such as torque, balancing, weight and cost will influence his decision. In the cost clause, attention will be given to details inseparable from multi-cylinder engines—number of sets of valve gears to use,

necessity of providing a crank axle, etc., and with each must be considered the increased maintenance charges as well as first cost. A further point will be that of crank angles, for while a four-cylinder unit will normally have adjacent cranks at 180 degrees apart, some additional advantages are obtainable by staggering the sequence as shown in Fig. 14, a softer and more regular blast assisting the boiler in producing a greater volume of steam. Where a three-cylinder engine is favoured, and it is found necessary to incline the inside cylinder and motion relative to the outside sets, the position of the inside crank must be corrected accordingly.

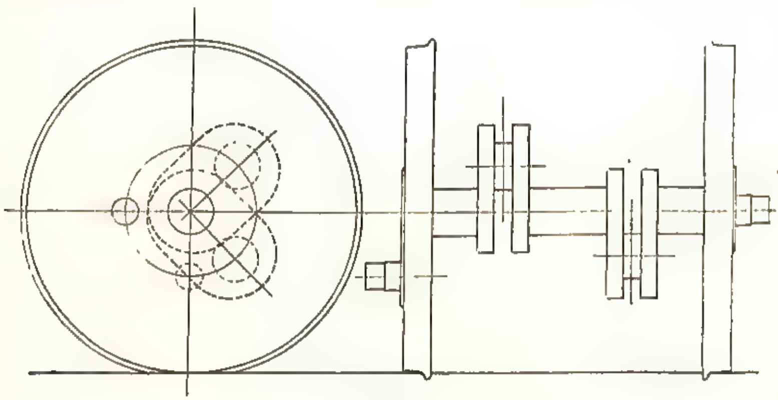


Fig. 14.

Steam Pressure.—Having the wheel diameter and the cylinder proportions decided, all that remains to be settled in order to complete all the factors in the T.E. formula for a simple engine is the boiler pressure, and this may be controlled by various parties. Even to-day a strong prejudice exists in many quarters against steam pressures over 160 pounds per sq. inch, yet many engines are doing fine work regularly on home railways at pressures well over 200 lbs. If the power required can be obtained by using a relatively low figure so much the better, as admittedly a low pressure does help in keeping down maintenance charges. But where a low pressure will handicap the success of an otherwise excellent design, raise the pressure to the level desired, and make the design in conformity, using as high a factor of safety as the weight limit will allow.

It is, of course, recognised that the boiler pressure and that in the steam chest immediately available for doing useful work, are by no means alike. A fall in pressure between these points is inevitable and the skilful designer will acquaint himself with the cause of such drop and proceed with the object of reducing this to the absolute minimum. A supply pipe or regulator which is

below size will adversely affect his machine's efficient operation; on the other hand, a multiple type regulator of the type frequently fitted to the headers of superheater locomotives will enhance his job, always assuming valves of suitable size are adopted. The design of the cylinders and valves, as well as the steam passages, will go a long way to make or mar the final result, and last, but by no means least, the boiler design must have the most careful attention to ensure the use of the full power of which the mechanism provided is capable.

There are factors that cannot be entirely eliminated in a normal design of locomotive which militate against the power output when speed enters into the calculation, as tests conducted some years ago in the U.S.A. by the Master Mechanics' Association showed that wire-drawing alone, where valves and steam passages of normal design are employed, accounts for a considerable drop in pressure in the cylinder, and this fact suggests the use of a special arrangement, such as the Caprotti, the Franklin or the Lentz, each of which employs the poppet type of valve, separate valves being carried for live and exhaust steam. Although this introduces a special feature, if results are the prime consideration, every alternative must be examined which offers an increased power output without recourse to a larger locomotive.

Where ordinary valve gears are to be used, Tables 5 and 6 below may be useful in arriving at approximate values for the mean pressure of steam available in the cylinder in percentages of the boiler pressure in relation to the revolutions per minute of the driving wheels (Table 5) and the approximate mean effective pressure available when related to the point of cut-off (Table 6).

TABLE 5.

Revs. per min.	60	80	100	120	140	160	180	200	220	240	260
Available M.P. in % of Boiler Pressures	86	76	66	58	51	46	42	39	34	32	30

TABLE 6.

a Period of admission in % of length of stroke.

p Mean pressure in % of maximum pressure of admission $p = 13{\cdot}5\sqrt{a} - 28$

a	17·5	20	25	30	35	40	45	50	55	60	65	70	75
p	28	32	40	46	52	57	62	67	72	77	81	85	89

(Table 5 is from a chart in the possession of the Master Mechanics' Association, U.S.A., and Table 6 is from "Railway Machinery," by D. K. Clark).

When the tractive power at a given speed is required, ascertain the mean pressure of steam at the speed stated, from Table 5. If, however, the maximum tractive force the engine will develop

is being considered, calculate the mean steam pressure by reference to Table 6.

There are in operation to-day in the U.S.A. and on the Continent, several locomotives employing an ultra-high pressure of steam, and the future alone can decide whether success will be achieved on these lines. Engines of somewhat similar design have been built and operated in Britain, but results so far have by no means justified the production of similar units in large numbers.

Heating Surfaces and Grate Area.—The proportioning of these areas suitably for the generation of steam in sufficient quantities for the mechanism to function to its fullest capacity involves a study too lengthy for the present article and the interested reader is referred to the pamphlet on "Locomotive Boiler Design" by the same writer.

Engine Weight.—Although a power/weight basis has been adopted in the field of motor and aero engines, no such rating seems practicable for locomotives and the factor of adhesion best suited for the local conditions under which the engine works must remain the determining value. If the engine power exceeds the correct proportion permitted by the adhesive weight, an excessive amount of slipping may arise, while if the weight is greater than is necessary for the power output, the locomotive will be sluggish and absorb an unreasonable percentage of energy in transporting its own mass.

Experience alone on the user's track will dictate the most desirable factor of adhesion for different services, and it is this value alone that must be applied to a new design. Cases exist where the designer deliberately evolved a high-power unit with a low adhesion weight in order that the reserve power could be called upon in times of emergency when sanding plus extra careful handling would assist in surmounting the critical point with little, or no, slipping.

When the locomotive proportions are finally cast, the next step will be to develop a full scheme arrangement embodying the leading features. From this an estimate of the total weight in running order can be made, and if this is acceptable, the distribution over the several axles will follow. This becomes purely one of re-arrangement where necessary to effect a satisfactory loading of each of the axles, and is almost always capable of solution. Should the total weight be too high, however, ways and means must be considered of reducing this to a figure which is acceptable to the C.M.E., and if the weight cannot be stepped down without seriously affecting the power output, it is usual to approach the Engineer for permission to employ a slightly heavier total than that originally contemplated.

Among the methods adopted for reducing the total weight, the following may be mentioned :—

(1) Special design of boiler which curtails the weight, while yet affording the maximum heating surfaces and steaming areas, etc., required. This results in a boiler and firebox structure tapering from a relatively small diameter at the smokebox tubeplate to the maximum across the firebox tubeplate, the firebox shell likewise tapering to a smaller width in plan and height in elevation at the backplate end.

(2) Nickel steel plates for boiler and firebox shell, and also for engine frame-plates.

(3) Steel inner firebox.

(4) Alloy steel for driving and coupling rods and valve gear rods.

(5) L.N.E.R. type piston and rod combined as one forging.

(6) Careful supervision of all structural and operating details, using stresses more in keeping with modern materials. Many of the stresses used to-day for the high-grade alloy steels chosen for locomotive construction are unduly conservative and would appear to have been influenced by stresses previously used, being little higher than those for plain carbon steel units, and seemingly disproportionate when compared with the yield values for the two classes of steel.

(7) Hollow axles, crank pins and piston rods.

It might be, however, that the total weight as estimated is acceptable, but the distribution is not, and steps are necessary to correct it. The trouble then is due to the position of the centre of gravity of the springborne masses—the C.G. may be too far forward or, more probably, too far behind. Engine designs with rear trucks are frequently troublesome in this respect, the trailing carrying unit being excessively loaded in many preliminary schemes. Several methods are available to achieve a successful issue—the introduction of a combustion chamber will throw the mass of tube and water weight so many inches forward and appreciably assist in the re-distribution. As the object before the designer is to move the wheelbase into its correct position relative to the upper structure, it becomes purely a drawing-board problem with the application of simple mechanics, and, as a rule, the latitude of movement available suffices to obtain a solution of the problem.

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7. Automobile Brakes (Transmission Brakes) } Connected.
8. Capacities of Bucket Elevators.
9. Valley Angle Chart for Hoppers and Chutes.
10. Shafts up to $5\frac{1}{2}$-in. diameter, subjected to Twisting and Combined Bending and Twisting.
11. Shafts, $5\frac{1}{2}$ to 26 inch diameter, subjected to Twisting and Combined Bending and Twisting.
12. Ship Derrick Booms.
13. Spiral Springs (Diameter of Rd. or Sq. Wire).
14. Spiral Springs (Compression).
15. Automobile Clutches (Cone Clutches).
16. ,, ,, (Plate Clutches).
17. Coil Friction for Belts, etc.
18. Internal Expanding Brakes. Self-Balancing Brake Shoes (Force Diagram).
19. Internal Expanding Brakes. Angular Proportions for Self-Balancing. } Connected.
20. Referred Mean Pressure Cut-Off, etc.
21. Particulars for Balata Belt Drives.
22. $\frac{7}{8}''$ Square Duralumin Tubes as Struts.
23. $1''$,, ,, ,, ,,
24. $\frac{3}{4}''$ Sq. Steel Tubes as Struts (30 ton yield).
25. $\frac{7}{8}''$,, ,, ,, (30 ,,).
26. $1''$,, ,, ,, (30 ,,).
27. $\frac{3}{4}''$,, ,, ,, (40 ,,).
28. $\frac{7}{8}''$,, ,, ,, (40 ,,).
29. $1''$,, ,, ,, (40 ,,).
30. Moments of Inertia of Built-up Sections (Tables)
31. Moments of Inertia of Built-up Sections (Instructions and Examples). } Connected
32. Reinforced Concrete Slabs (Line Chart)
33. Reinforced Concrete Slabs (Instructions and Examples) } Connected
34. Capacity and Speed Chart for Troughed Band Conveyors.
35. Screw Propeller Design (Sheet 1, Diameter Chart)
36. ,, ,, ,, (Sheet 2, Pitch Chart)
37. ,, ,, ,, (Sheet 3, Notes and Examples) } Connected
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39. Close Coil, ,, ,,
40. Trajectory Described by Belt Conveyors.
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42. Useful Conversion Factors.
43. Torsion of Non-Circular Shafts.
44. Railway Vehicles on Curves.
45. Chart of R.S. Angle Purlins.
46. Coned Plate Development.
47. Solution of Triangles (Sheet 1, Right Angles).
48. Solution of Triangles (Sheet 2, Oblique Angles).

49. Relation between Length, Linear Movement and Angular Movement of Lever. (Diagram and Notes).
50. ,, ,, ,, ,, ,, ,, (Chart).
51. Helix Angle and Efficiency of Screws and Worms.
52. Approximate Radius of Gyration of Various Sections.
53. Helical Spring Graphs (Round Wire)
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56. Relative Values of Welds to Rivets.
57. Ratio of Length/Depth of Girders for Stiffness.
58. Graphs for Strength of Rectangular Flat Plates of Uniform Thickness.
59. Graphs for Deflections of Rectangular Flat Plates of Uniform Thickness.
60. Moment of Resistance of Reinforced Concrete Beams.
61. Deflection of Leaf Spring.
62. Strength of Leaf Spring.
63. Chart showing relationship of various Hardness Tests.
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68. Hub Pressed on to Steel Shaft. (Radial Gripping Pressure between Hub and Shaft).
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82. Steam Pipe Design. Chart showing flow of Steam through Pipes.
83. Variation of Suction Lift and Temperature for Centrifugal Pumps.
84. Nomograph for Uniformly Distributed Loads on British Standard Beams.
85. ,, ,, ,, ,,
86. ,, ,, ,, ,, } Connected
87. Notes on Beam Design and on Use of Data Sheets Nos. 84-5-6.
88. ,, ,, ,, ,,

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